Renewable and Sustainable Energy Reviews 16 (2012) 6647–6659

Contents lists available at SciVerse ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Ejector enhanced vapor compression refrigeration and heat pump systems—A review

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ARTICLE INFO

Article history:
Received 21 April 2012
Received in revised form
25 July 2012
Accepted 4 August 2012
Available online 5 October 2012

Keywords:
Refrigeration and heat pump
Vapor compression system
Cycle configuration
Two-phase ejector
Performance characteristic

ABSTRACT

The present paper provides a literature review on two-phase ejectors and their applications in vapor compression refrigeration and heat pump systems. Geometry, operation and modeling of ejector, and effects of various operating and geometric parameters, and refrigerant varieties on the ejector performances as well as performance characteristics of both subcritical and transcritical vapor compression systems with various cycle configurations are well-summarized. Moreover, system optimal operation and control to get maximum performance by using ejector as an expansion device are also discussed. However, a lot of research work still needs to be done for large-scale applications in industry and for the replacement/modification of conventional refrigeration and heat pump machines. Favorable performance improvement along with several advantages in installation, operation and control with ejector stimulates the commercialization of ejector enhanced refrigeration and heat pump systems and hoping this contribution will be useful for any newcomer in this field of technology.

Contents

1. Introduction

Performance enhancement of refrigeration and heat pump systems by cycle modification is an emerging research topic now-a-days to reduce the electricity consumed leading to mitigating the problems related to the environmental pollution by utility power plants. Due to no moving parts, low cost, simple structure and low maintenance requirements, the use of two-phase ejector has become a promising cycle modification recently. The main advantage of the ejector may be found in the recovery of the expansion work normally wasted in throttling processes at a typical expansion valve. Use of ejector as an

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http://dx.doi.org/10.1016/j.rser.2012.08.007

Nomenclature

a	cross-sectional area, m^2
COP	coefficient of performance
h	specific enthalpy, J/kg
$\dot{m}$	mass flow rate, kg/s
p	pressure, Pa
s	specific entropy, J/kg K^{-1}
u	fluid velocity, m/s
Δp_f	frictional pressure drop, Pa
η_e	ejector efficiency
η_{is}	isentropic efficiency
μ	entrainment ratio
ρ	fluid density, kg/m^3

expansion device by replacing the throttling valve in the vapor compression refrigeration cycle seems to be one of the efficient ways to reduce the throttling losses or the expansion irreversibility in the refrigeration/heat pump cycle. Ejector also reduces the compressor work by raising the suction pressure to a level higher than that in the evaporator leading to the improvement of COP. Use of ejector will give two benefits: work recovery (COP improvement) and flash gas bypass (evaporator size reduction).

Two-phase ejectors have constituted an attractive alternative for classic expansion devices for several decades. The steam jet refrigeration system was first developed in early 1900s and some review papers have been published to summarize the research efforts and achievements focused on ejector modeling and performance optimization, and applications in refrigeration [1–4]. However, the use of ejector in vapor compression system for performance improvement was first proposed in the 1990s by Kornhauser [5], who analyzed the thermodynamic performance of the ejector expansion refrigeration cycle using R12 as a refrigerant. After his work, huge researches have been conducted on this field within last two decades. However, to the best of the authors' knowledge, only one review [6] is published recently, which is only limited to operation principle, thermodynamic modeling of single stage refrigeration cycle and effects of operating parameters and working fluids on performance, but not on ejector enhanced advanced cycle configurations and the effects of design parameters on both ejector and system performances and control.

The aim of this paper is to provide basic background knowledge and a review of existing literatures on two-phase ejectors and their applications in vapor compression system. Various ejector enhanced systems and operating characteristics are well-grouped and it is hoped that this paper will be useful for any newcomer and manufacturers in this field of refrigeration and heat pump technologies.

2. Ejector operation, modeling and performance

An ejector is also known as jet, injector or jet pump in different literatures. The main components of an ejector include a primary nozzle (also named as motive nozzle), the suction chamber, the mixing chamber (including a convergent chamber if available and a constant-area throat tube) and the diffuser (Fig. 1). The primary nozzle may be a convergent type or a convergent–divergent type. As the high pressure fluid, known as primary fluid or motive fluid, expands and accelerates through the primary nozzle, it flows out with high speed to create a very low pressure region at the nozzle exit plane. Hence, a pressure difference between the streams at the nozzle exit plane and the secondary fluid inlet is established and a secondary fluid is drawn through suction chamber or nozzle by the entrainment effect. Then both fluids mix in the mixing chamber and flow through the diffuser to convert the kinetic energies of mixture to pressure energy. Therefore, the mathematical description of the flow inside the ejector is very complex. Besides the conservation equations of mass, energy and momentum, state equations, phase change principle, isentropic relations as well as some appropriate assumptions need to be used to assist in the description of the flow and mixing in the two-phase ejector [4].

Normally, the ejector design can be classified into two types according to the position of the nozzle [1]. The ejector, which has the nozzle with its exit plane located within the suction chamber in front of the constant-are section and the static pressure is assumed to be constant through the mixing process is known as a constant-pressure mixing ejector. For the nozzle with its exit located within the constant-area section, the ejector is called a constant-area mixing ejector. It was found that the constant-pressure mixing ejector had a better performance than the constant-area one [1]. Although, compared to constant-pressure mixing, the theory of constant-area mixing was much better understood and performance could be predicted with a much greater degree of accuracy [2]. The phenomenon of choking is usual when supersonic flow occurs and limits the mass flow of the secondary and primary flows. In practice, two choking phenomena exist in the ejector performance [7]: one in the primary flow through the nozzle and the other in the entrained flow. Huang et al. [7] analysed choking condition with constant pressure mixing and showed that 1-D analysis using the empirical coefficients could accurately predict the performance of the ejectors. However, the 1-D assumption is not able to describe the actual velocity distribution [4]. To make the simulated model become more realistic, the isentropic efficiency including the friction losses can be take in to account, which is dependent on fluid properties, ejector geometries and operating conditions. Non-isentropic flow in nozzles and diffuser sections, mixing, flow friction and shock are the causes of ejector irreversibility. Flow visualization and static wall pressure distributions indicate the

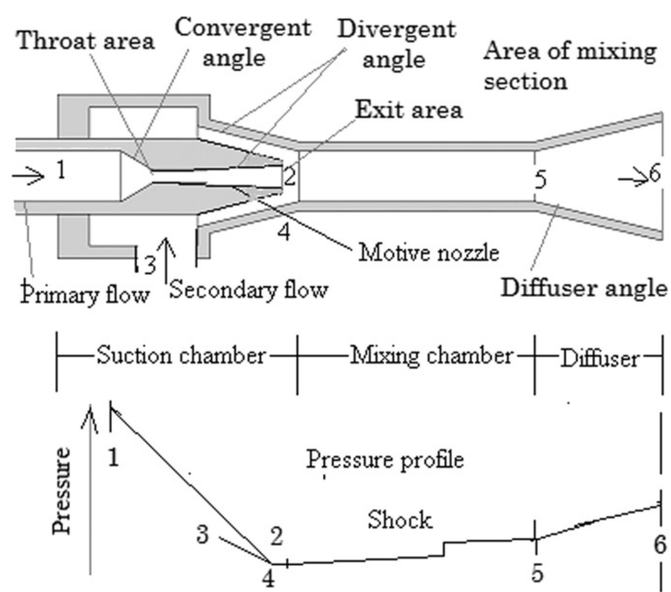

Fig. 1. Ejector geometry and pressure distribution.

existence of complicated two-phase shock patterns in the mixing section with a significant impact on ejector performance [8]. Furthermore, the non-homogenous formulation of a two-phase fluid is more realistic approach, which requires deep knowledge of the nature of flow and clearly defined flow regimes for the two-phase ejectors. However, in some cases the homogeneous flow model could offer a reasonable substitute and avoiding the complexity of heterogeneous model, empirical/semi-empirical model can also be used for two-phase ejector [4]. Nakagawa et al. [9] experimentally proved that the decompression pressure profiles for a diverging section of the CO_2 motive nozzles may significantly differ from the profiles calculated according to the Isentropic Homogeneous Equilibrium (IHE), which indicates boiling processes started at non-equilibrium supersaturated state and hence the homogeneous equilibrium approach is not suitable for the motive nozzle calculations, neither design nor off-design type. Hence, Banasiak and Hafner [10] recently utilized the Delayed Equilibrium Model supplied with the Homogeneous Nucleation Theory for the purpose of the metastable states analysis for a transcritical CO_2 flow with delayed flashing over the motive nozzle. Recently, some CFD analyses have been done for transonic and supersonic two-phase ejector to take care of complex physics involving nonequilibrium thermodynamics, shear mixing, and void fraction-dependent speed of sound [11,12].

According to one-dimensional homogeneous equilibrium model (mainly used recently for two-phase ejector simulation), the following mathematical model for ejector (Fig. 1) per unit total mass flow rate can be established based on further assumptions of (i) Both the motive stream and the suction stream reach the same pressure at the inlet of the mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section, (ii) the expansion efficiencies of the motive stream and suction stream are given constants. The diffuser of the ejector also has a given efficiency. Any shock effects are included in these efficiencies, (iii) kinetic energies of the refrigerant at the ejector inlet and outlet are negligible, (iv) homogeneous equilibrium flow conditions are considered at the nozzle outlet in the primary flow, (v) the ejector is adiabatic, rigid and impermeable.

Energy balance equations in the primary and secondary nozzle sections:

$$h_1 = h_2 + \frac{u_2^2}{2}, \quad \eta_{is} = \frac{h_1 - h_2}{h_1 - h[p_2, s_1]} \tag{1}$$

$$h_3 = h_4 + \frac{u_4^2}{2}, \quad \eta_{is} = \frac{h_3 - h_4}{h_3 - h[p_4, s_3]} \tag{2}$$

Mass equations in term of entrainment ratio ($\mu = \dot{m}_4/\dot{m}_2$):

$$\rho_2 a_2 u_2 = \dot{m}_2 = 1/(1+\mu) \tag{3}$$

$$\rho_4 a_4 u_4 = \dot{m}_4 = \mu/(1+\mu) \tag{4}$$

Mass, momentum and energy equations in the mixing section:

$$\rho_2 a_2 u_2 + \rho_4 a_4 u_4 = \rho_5 a_5 u_5 = 1 \tag{5}$$

$$p_2 a_2 + \frac{1}{1+\mu} u_2 + p_4 a_4 + \frac{\mu}{1+\mu} u_4 = p_5 a_5 + u_5 + \Delta p_f a_5 \tag{6}$$

$$\frac{1}{1+\mu}\left(h_2 + \frac{u_2^2}{2}\right) + \frac{\mu}{1+\mu}\left(h_4 + \frac{u_4^2}{2}\right) = h_5 + \frac{u_5^2}{2} \tag{7}$$

Although the values of cross-sectional areas of nozzle exits and mixing section depends on ejector design, for simplicity, one can take $a_5 = a_2 + a_4$. It may be noted that to avoid the complexity of two-phase mixing, the pressure drop has been neglected for most of the theoretical studies and sometimes, mixing efficiency has been considered in account to pressure drop which is dependent on ejector geometry and operating conditions.

The energy balance in the diffuser section:

$$h_6 = h_5 + \frac{u_5^2}{2}, \quad \eta_{is} = \frac{h[p_6, s_5] - h_5}{h_6 - h_5} \tag{8}$$

The overall energy balance in the ejector:

$$\frac{1}{1+\mu} h_1 + \frac{\mu}{1+\mu} h_3 = h_6 \tag{9}$$

For refrigeration applications, the most important performance parameters of ejector are an entrainment ratio or circulation ratio and a pressure lift ratio ($=p_6/p_3$), which are related to the cooling capacity and compressor work. Therefore, there is no doubt that an ejector operating at the given operating conditions with the highest entrainment ratio and highest pressure lift ratio will be the most desired ejector [1]. These parameters were also found to directly depend on ejector geometries and its working fluid. Many recent studies proposed and tested some new criterion in designing an ejector which has higher pressure lift performance. These ideas were to minimize any losses created by a mixing and shocking process. Another parameter, ejector efficiency (ratio of actual work recovered by the maximum work recovery potential) is given by [13],

$$\eta_e = \frac{(\dot{m}_2 + \dot{m}_4)\Delta h_c - \dot{m}_4 u_4^2/2}{\dot{m}_2 (h_1 - h_2)} = \frac{(1+\mu)\Delta h_c - \mu u_4^2/2}{(h_1 - h_2)} \tag{10}$$

where, Δh_c is the work saved for compressing from evaporator pressure to compressor inlet pressure.

Sizing an ejector consists of nozzle position, throat diameter of the nozzle, outlet diameter of the nozzle, constant area diameter of the mixing chamber, diffuser outlet diameter, divergent and convergent angles of nozzle, suction chamber converging angle, diffuser diverging angle and the constant area chamber length. These geometric parameters will affect the ejector's performance. According to the invention by Takeuchi et al. [13], the ratio of mixing portion diameter to nozzle exit diameter is in a range of 1.05–10, the ratio of length to diameter of mixing portion is equal to or smaller than 120, the length of diffuser is 10–14 times of mixing length and an extension angle of diffuser is in a range of 0.2–34°. However, these parameters can be optimized to get maximum ejector efficiency [13], which are dependent on operating conditions and refrigerant used.

Use of ejector in vapor compression system will ultimately affect the system performance parameters such as cooling or heating capacity and COP, volumetric cooling or heating capacity, exergy losses (irreversibilities) of system and its components, and second law efficiency.

3. Single stage compression cycle with ejector

3.1. Subcritical cycle configurations and operation

Kornhauser [5] first analyzed the ejector expansion refrigeration cycle using R12 as a refrigerant. The layout of ejector expansion vapor compression refrigeration cycle is shown in Fig. 2. The principal modifications from the standard refrigeration system are the addition of a two-phase ejector and a liquid–vapor separator. The primary flow from the condenser (state 3) and the secondary flow from the evaporator (state 8) are going through primary and secondary nozzles, respectively, constant area mixing and diffuser (1 0 –5) sections of the ejector and then separated in forms of vapor (state 1) and liquid (state 6) so that this ratio should matched with the inlet ratio of primary and secondary flows. Then the liquid circulates through expansion

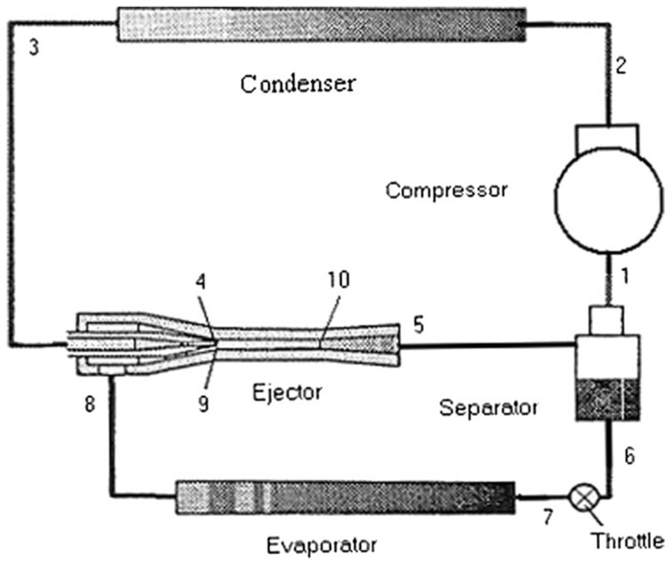

Fig. 2. Ejector expansion vapor compression subcritical cycle.

valve (6–7) and evaporator (7–8), whereas the vapor circulates through compressor (1–2) and condenser (2–3). The throttle valve is used on for small pressure difference and thus causes little expansion losses [5]. The two-phase ejector in the refrigeration cycle enables the evaporator to operate as in a liquid recirculation system. The theoretical and experimental studies have been conducted with various working fluids and one dimensional homogeneous model has been used for the ejector performance analysis.

3.2. Subcritical cycle performance characteristics

Kornhauser [5] analyzed the ejector expansion refrigeration cycle using R12 as a refrigerant based on constant mixing pressure model and found a COP improvement of up to 21% over the standard cycle under standard operating conditions. Initial experimental study [14] showed that the performance of two phase ejector is inferior to single phase ejector and COP improvements of refrigeration cycle ranging from 3.9% to 7.6% with R134a as a refrigerant. Their further testing after some modifications showed significant performance improvement [15]. Domanski [16] found that the theoretical COP of the ejector expansion refrigeration cycle was very sensitive to the ejector efficiency. Bilir and Ersoy [17] showed that even in the case of an off-design operation, the performance of a system with ejector is higher than that of the basic system. Yari [18] has done the second law analysis of ejector expansion vapor compression refrigeration cycle using R134a as a refrigerant. For typical air-conditioning application, the total exergy destruction of the vapor compression cycle is about 24% higher than that for the ejector-compression cycle and the second law efficiency of the ejector-compression cycle is about 16% higher than that for the vapor compression cycle [18]. Second law performance is strongly dependent on isentropic efficiencies of motive nozzle, suction nozzle and diffuser [19]. An exergetic analysis showed that introduction of two ejectors in cascade system ameliorate the exergy efficiency by 27.3% when CO_2 is used in lower and R-152a in higher sides for evaporator temperature of $-30\,^{\circ}C$ [20].

Disawas and Wongwises [21] experimentally investigated the performance of ejector expansion refrigeration cycle without the expansion valve up stream of the evaporator so that the evaporator is flooded with the refrigerant. Their test showed the COP improvement over the conventional cycle at low heat sink temperature with R134a as the refrigerant. The motive mass flow rate in the ejector was found to be strongly dependent on the heat sink temperature and independent of the heat source temperature. The system enables the evaporator to be flooded with refrigerant, resulting in a higher refrigerant-side heat transfer coefficient [22]. The cooling capacity and COP increase with the rise of the heat source temperature due to increase in entrainment ratio, and decrease with increase in heat sink temperature due to decrease in entrainment ratio [23]. The relative COP increases with decrease in evaporator temperature for synthetic refrigerants [23]. Study by Bilir and Ersoy [17] for R134a using constant area ejector showed that as the difference between condenser and evaporator temperatures increases, the improvement ratio in COP rises. Studies by Sarkar [24,25] also showed that the COP improvement over basic expansion cycle increases due to increase in pressure lift ratio with the increase in condenser temperature and decrease in evaporator temperature. Experimental study [26] showed that the ejector efficiency is also dependent on the operating conditions.

The performance improvement by using ejector as an expansion device is strongly dependent on nature of working fluids also. Using ejector as an expansion device, R134a yields better performance improvement compared to R404A, R407 and R410 [13]. Nehdi et al. [27] compared several synthetic refrigerants and best performance of 22% COP improvement was obtained with R141b. Sarkar [24,25] compared three natural refrigerants and showed maximum performance improvement by using ejector can be achieved in case of isobutane, whereas minimum performance improvement can be achieved for ammonia. Further the values of optimum area ratio and corresponding entrainment ratio and pressure lift ratio are also dependent on refrigerant used [24,25]. Fong et al. [28] simulated solar-electric driven ejector-assisted vapor compression chiller for space conditioning in subtropical climate using R22, R134a and R410A, and showed that the coefficient of performance of the chiller was increased and the total primary energy consumption of the system was decreased for all the three refrigerants, in which the degree of enhancement from R134a was the most significant.

Nakagawa and Takeuchi [29] showed that the longer the length of the divergent part of the motive nozzle, the higher the motive nozzle efficiency could be achieved leading performance improvement. This was likely caused because the longer divergent part provided a longer period of time for the two-phase flow to achieve equilibrium. With the increase in primary nozzle throat diameter, the recirculation ratio was found to be increased and both cooling capacity and COP were found to be increased whereas effect on compressor pressure ratio was found to be negligible [23]. However, the effect of outlet diameter of motive nozzle is negligible on the system performances [30].

The effect of the area ratio (mixing section to motive nozzle exit) on the COP is not monotonous. It has been shown that for fixed condenser and evaporator temperatures of the ejector expansion cycle, the COP increases until a maximum value is reached and then decreases with increasing area ratio. The change in COP is associated with the change of the compressor suction pressure, when the compression suction pressure increases, the load on the compressor decreases, and conversely. Therefore, there exists an optimum area ratio, which means that the system has maximum COP. If the ejector operates beyond optimum value, some energy is wasted and consequently the compression ratio increases and the COP decreases. Nehdi et al. [27] theoretically investigated the performance of ejector expansion refrigeration cycle with several synthetic refrigerants and showed that the optimum area ratio is around 10. However, for the different operating temperatures there are different optimum values of pressure drop in the suction chamber, ejector area ratio, ejector outlet pressure and

corresponding maximum COP. As the difference between condenser and evaporator temperatures increases, the optimum ejector area ratio drops [17]. Studies by Sarkar [24,25] for natural refrigerants by using both constant pressure and constant area mixing ejectors also showed that optimum area ratio increases with increase in evaporator temperature and decrease in condenser temperature. Author has developed the expressions for optimum ejector geometric parameters for ammonia, propane and isobutane as guidelines for optimal design and for selecting appropriated operating conditions. Optimum entrainment ratio increases towards the minimum condenser temperature and maximum evaporator temperature, whereas the optimum pressure lift ratio increases towards the maximum condenser temperature and minimum evaporator temperature [24,25]. It was also found that optimum ejector area ratio increases with decrease in ejector component efficiencies [19].

3.3. Transcritical cycle configurations and operation

Practical implementation of two-phase ejector in transcritical CO_2 refrigeration or heat pump system is quite easier due to relatively small expansion ratio compared to the conventional refrigeration system. Liu et al. [31] first proposed the ejector expansion transcritical CO_2 refrigeration cycle. The layout of the conventional ejector-driven transcritical CO_2 cycle is shown in Fig. 3(a). The primary flow from the gas cooler and the secondary flow from the evaporator are passing through nozzle, mixing and diffuser sections of the ejector and then separating in forms of vapor and liquid so that this ratio should matched with the inlet ratio of primary and secondary flows. The ejector entrainment ratio is equal to the mass ratio of the two streams in a stable system while the mass percent of saturated vapor is equal to the vapor quality at the ejector exit. It can be seen that the quality of state point 1 is fixed at one and the quality of state point 6 is fixed at zero. Thus, the entrainment ratio of the ejector, μ, and the quality of the ejector outlet stream at state point 5, x_5, has to satisfy $x_5=1/(1+\mu)$ to meet the mass conservation constraint for steady-state operation of the cycle. However, for a given ejector configuration, the entrainment ratio of the ejector is determined by the motive flow and suction flow and the ejector outlet pressure. This leads to a difficulty to control the operating conditions of a real system. To relax the constraints between the entrainment ratio of the ejector and the quality of the ejector outlet stream, a new ejector expansion transcritical CO_2 refrigeration cycle (Fig. 3(b)) was proposed by Li and Groll [32]. Part of the vapor in the separator is feed back to the evaporator inlet through a throttle valve, which regulates the quality at the evaporator inlet. The throttle valve can be controlled by the liquid

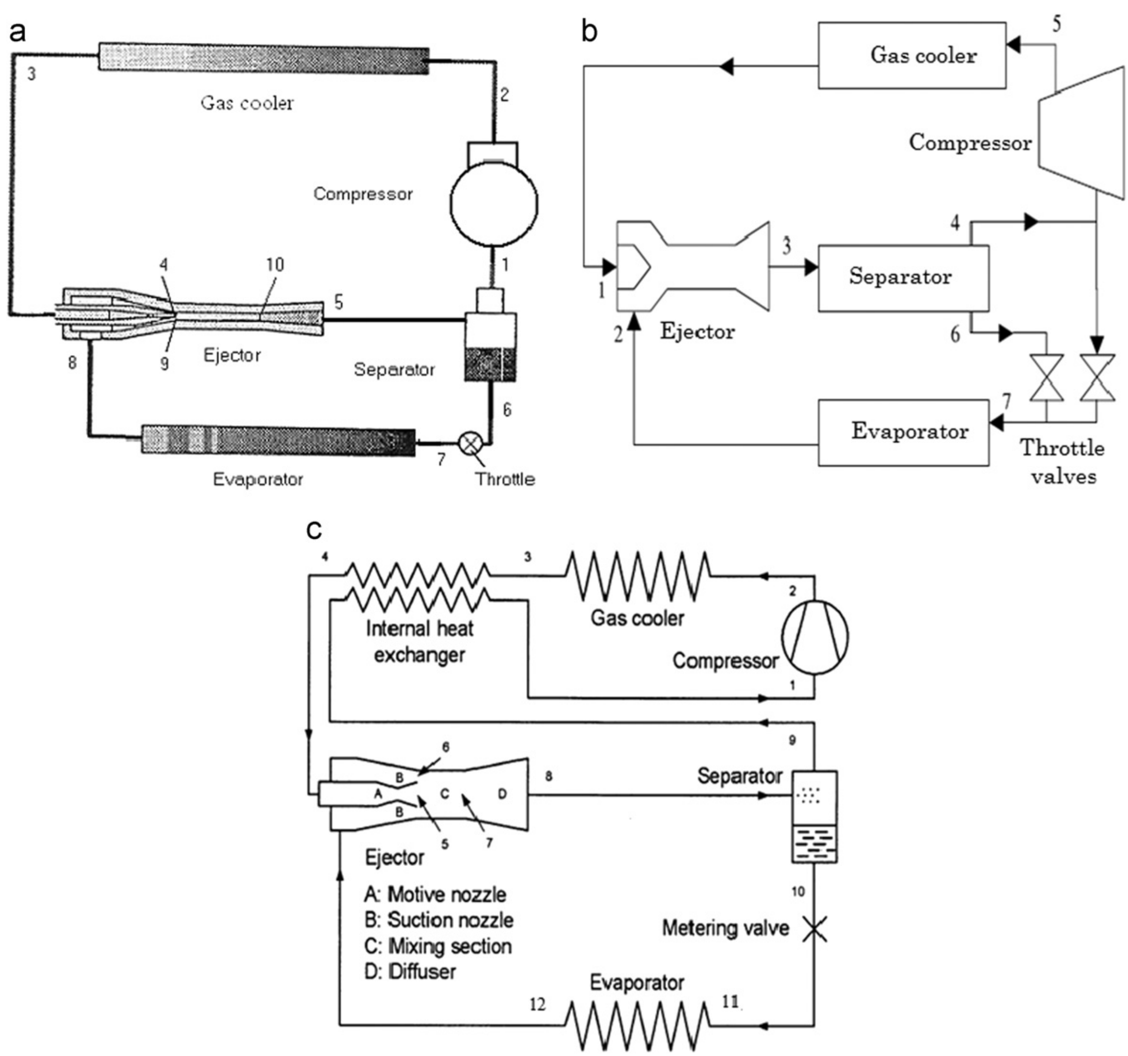

Fig. 3. Ejector expansion vapor compression transcritical CO_2 cycle.

level in the separator to ensure that the mass conservation is being satisfied to maintain a steady-state operation. Ejector-expansion transcritical CO_2 refrigeration cycle with internal heat exchanger [33] is shown in Fig. 3(c). A suction line heat exchanger is used in ejector expansion cycle to get superheating (9–1) before compressor and subcooling (3–4) after gas cooler. Cogswell et al. [34] recently patented single-stage ejector expansion cycle (similar to Fig. 3(a)) and ejector expansion cycle with suction line heat exchanger for subcooling of separator liquid using superheating of evaporator exit vapor (here, the suction heat exchanger is used in suction flow side, whereas this is used in primary flow side in case of Fig. 3(c)) for dual mode operations i.e., with or without ejector.

One-dimensional homogeneous equilibrium flow model with both constant pressure and constant area mixings have been mainly used for the theoretical analysis. It is well-known that the gas cooler pressure control is an important task for transcritical cycle to get optimum performance. The gas cooler pressure can be controlled by changing throat area of nozzle since the flow rate is proportional to throat area of nozzle [35]. In the ejector expansion cycle, due to significantly lower pressure difference in throttling, the mass flashed into vapor is greatly reduced and the two-phase flow can be more easily distributed. Therefore, the evaporator tube arrangement is not as critical in the ejector expansion cycle [36]. The metering valve installed downstream of the vapor–liquid separator can be used to control the balance between ejector entrainment and pressure lift [37]. This allows compensation of effects caused by operation at different ambient temperatures and at off-design compressor speeds. Literature showed that some geometric parameters, which can be adjustable during operation for certain ejector design, are motive nozzle throat and exit areas (by using needle) and suction nozzle exit area by adjusting distance between motive nozzle exit and entry of mixing section (motive nozzle position). These parameters give three dimensionless parameters: ratio of mixing area to motive nozzle throat area, area ratio of motive nozzle exit to throat and area ratio of suction nozzle exit to motive nozzle exit. Motive nozzle throat and exit areas can be adjusted by means of a moving nozzle needle successfully to get the maximum system performance [38].

3.4. Transcritical cycle performance characteristics

Through the thermodynamic analysis, Liu et al. [31] first showed the performance improvement of the transcritical CO_2 refrigeration cycle using ejector. The COP improvement by using ejector in transcritical CO_2 cycle is more significant compared to subcritical cycle [33]. Significant COP improvement has been observed by other studies also [32]. However, the unavoidable losses caused by irreversible mixing in ejector can degrade the system performance [35]. Ksayer and Clodic [39] theoretically showed that the COP of the ejector expansion transcritical CO_2 cycle can be improved by more than 15% compared to the conventional transcritical cycle for typical air-conditioning operating conditions. Deng et al. [36] showed that the ejector expansion system maximum cooling COP is up to 18.6% better than the internal heat exchanger cycle cooling COP and 22.0% better than the conventional vapor compression refrigeration cycle cooling COP. The COP of the ejector expansion transcritical CO_2 cycle can be improved by more than 15% compared to the basic transcritical cycle for typical air-conditioning operating conditions [40]. Similar improvement has been also observed for the experiment by Lee et al. [41]. However, ejector expansion cycle can give maximum performance for certain range of pressure lift [42]. Nakagawa et al. [43] reported a COP improvement of up to 35%. Elbel [8] reported 9% cooling capacity improvement by using ejector. Banasiak et al. [44] reported maximum increase in the coefficient of performance (COP) of 8% over a system with a conventional expansion valve. In a CO_2 air-conditioning system with a constant speed compressor, both COP and cooling capacity enhancements became more significant as the outdoor air temperature increased (COP increased by up to 36%) and the compressor frequency decreased (COP increased by up to 147%) [45].

In the transcritical CO_2 ejector expansion cycle, the throttling exergy loss is much less than in the internal heat exchanger cycle and in the conventional cycle [36]. The exergy losses due to the compression and heat rejection processes are also reduced some. The evaporation exergy loss is not changed much [36]. Hence, the second law efficiency improves by using ejector in transcritical CO_2 cycle, although it is less than by using expansion turbine [46]. However, the theoretical analysis by Ersoy and Bilir [47] showed that the total irreversibility of the ejector system was lower than those of the basic and turbine expander systems by 39.1% and 5.46%, respectively under certain conditions. Another study [48] showed that ejector instead of throttling valve can reduce more than 25% exergy loss and increase more than 30% COP. Not only single system, the ejector can improve the performance in cascade system also [49].

Operating conditions affect the performance of ejector enhanced system. The COP improvement by using ejector in transcritical CO_2 cycle will be reduced with increase in evaporator outlet superheat [32]. Unlike to the conventional system, the optimum gas cooler pressure increases with evaporator temperature [36]. The ejector entrainment ratio significantly influences the refrigeration effect with an optimum ratio giving the ideal system performance [36]. Through the theoretical analysis, Sarkar [46] showed that the entrainment ratio, pressure lift ratio and COP at optimum gas cooler pressure are dependent on operating temperatures, compressor efficiency and ejector parameters (suction pressure drop, nozzle and diffuser efficiencies), which are dependent on component design. The optimum entrainment ratio increases with increase in evaporator temperature [36,40,46] and decreases with increase in gas cooler exit temperature [36,46], whereas the optimum pressure lift ratio increases towards the maximum gas cooler exit temperature and minimum evaporator temperature [46]. The modified cycle layout (Fig. 3(b)) can be realized for certain discharge pressure and entrainment ratio combinations. Conventional one (Fig. 3(a)) is always better than modified one (Fig. 3(b)) in term of system COP as well as cost due to lower optimum discharge pressure [46]. As discussed, optimum conditions can be maintained by using adjustable ejector [38]. The suction nozzle pressure drop had a significant effect on the ejector area ratio and performance and hence it is necessary to design an ejector with the optimum area ratio to achieve the optimum suction pressure drop and maximum performance [47]. Lucas and Koehler [50] experimentally optimized the high side pressure and showed that COP improvements of the ejector cycle of 17% were reached with ejector efficiencies of up to 22% compared to the maximal COP of the expansion valve cycle. It may be noted that their experimental values of optimum high side pressure to get maximum COP for various evaporator and gas cooler exit temperatures fairly match with the predicted data from correlation proposed by Sarkar [46]. Xu et al. [51] used an adjustable ejector subjected to electrical pulses through the stepper motor which drives needle forward or backward movement to change the nozzle throat area for the optimum high-side pressure control. They showed that the ejector high-side pressures for maximum COP are lower than that for maximum capacity.

Ejector as well as system performances are significantly affected by ejector geometry. With the increase in nozzle and diffuser efficiencies, entrainment ratio insignificantly increases and performance increases [36]. The pressure ratio of the motive

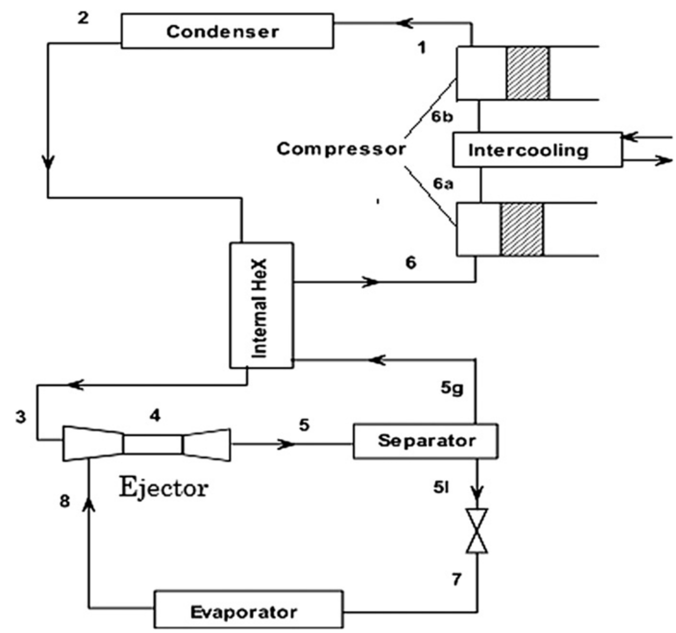

Fig. 4. Ejector expansion multistage compression subcritical cycle.

nozzle inlet to suction nozzle inlet increases as the motive nozzle throat diameter decreases, which results in a decrease of the motive nozzle efficiency [52]. With the increase in constant area diameter, the COP first increases, and gives maximum value and then decreases whereas optimum gas cooler pressure and corresponding entrainment ratio remains nearly constant [32]. Experimental study [37] showed that the diffuser angle has significant influence on static wall pressure distribution or pressure lift. The distance from the motive nozzle exit to the mixing section constant area entry not only affects the suction nozzle efficiency, but also affects the mixing section efficiency [52]. Liu and Groll [52] optimized the motive nozzle throat diameter and the mixing section diameter to get maximum pressure lift. The diffuser exit pressure increases quickly and then slowly as a function of the diffuser exit to mixing section diameter ratio [52]. Mixing cross-sectional area has also significant effect on performance [43]: smaller mixing area yields higher efficiency due to its higher pressure recovery and entrainment ratio, but its advantages are limited to lower ejector inlet pressure, whereas larger mixing area is required for higher cooling capacity which can be achieved at higher ejector inlet pressure or lower ejector inlet temperature but excessive increase in this area considerably decreases the efficiency of the system. Nakagawa et al. [43] experimentally showed 10% COP difference between smallest and largest mixing areas. An experimental result [53] showed that the mixing length had significant effect on entrainment ratio and on magnitude and profile of pressure recovery. The higher mixing length yielded the highest pressure recovery, suction flow rate, entrainment ratio, ejector efficiency and COP in all of the conditions used by Nakagawa et al. [53]. A COP improvement of up to 26% over conventional system was obtained but improper sizing of mixing length lowered the COP by as much as 10%. A much longer mixing length would have minor change in pressure recovery but it yields significant penalty in entrainment ratio. COP reached its maximum when the motive nozzle throat is chocked and the mass flow rate in suction nozzle is a maximum [41]. Lee et al. [41] experimentally showed that there exist also optimum mixing section diameter and optimum distance between motive nozzle and diffuser to get maximum COP for air-conditioning application. The considerable influence of friction forces on the mechanism of pressure recovery indicated that the length optimization procedure might be crucial for proper design [44]. Banasiak et al. [44] reported that the highest values of the ejector efficiency can be obtained for diffuser divergence angle of 3–5°, mixing length of 20–25 mm and mixing diameter of 3–3.5 mm. Ejector efficiency decreases after optimum mixing length due to significant pressure drop caused by friction forces occurred along the duct and also decreases after optimum diameter due to the poorer momentum exchange, possibly as a result of intensified recirculation. They also showed that the better performance can be obtained for the larger outlet diameter of the diffuser. Liu et al. [45] showed that ejection ratio and COP can increase significantly as ejector motive nozzle throat diameter or compressor frequency decreases and the entrainment ratio, pressure lift ratio, cooling capacity and COP reached maxima level when the distance between ejector motive nozzle exit and mixing section constant-area entry was equal to 3 times the mixing section diameter. They [54] also showed that cooling capacity and COP reach maximum values when motive nozzle throat diameter becomes 2.8 mm and a mixing section constant-area diameter of between 4.1 and 4.2 mm. The ejector geometry has to be adjusted as the operating conditions change, or a performance penalty of the ejector expansion transcritical CO_2 systems may result [54].

Effect of using internal heat exchanger in ejector-expansion transcritical CO_2 refrigeration cycle is not significant. Study [33] showed the improvement of COP by using internal heat exchanger for ejector-expansion system is lower compared to conventional system. Optimum high side pressure can reduce by using internal heat exchanger [33]. Unlike to the subcritical cycle, effect of mixing pressure (dependent on area ratio) on COP of ejector-expansion cycle with internal heat exchanger is not significant [33]. It was also shown that the cooling capacity and COP simultaneously improved by up to 8% and 7%, respectively for the air-conditioning application [55]. The prototype ejector was able to recover up to 14.5% of the throttling losses. Smallest diffuser angle of 5° yielded best results for the static pressure recovery of the high-speed two-phase flow entering the diffuser. Experiments confirmed that like in a conventional system with expansion valve, the high-side pressure control integrated into the ejector could be used to maximize the system performance. Experimental study by Xu et al. [56] also showed that the performance of ejector expansion CO_2 cycle degrades by using internal heat exchanger. The primary flow rate increases by using internal heat exchanger and hence heating capacity increases. The pressure lift and actual ejector work recovery are reduced when the internal heat exchanger is added to the ejector system. Their calculation showed that the compression ratio is reduced by 10.0–12.1% using ejector, while that is reduced only by 5.6–6.7% using both ejector and internal heat exchanger compared to that of a conventional CO_2 system. The entrainment ratio and the pressure lift increase with a decrease in cooling water flow rate or an increase in the inlet water temperature [56]. Unlike to other studies, experimental results by Nakagawa et al. [57] showed that internal heat exchanger significantly increased the COP of ejector system. At the conditions used in their research, the ejector system with 60 cm internal heat exchanger provided the maximum COP improvement of up to 27% compared to similar conventional system. The motive nozzle's inlet condition had significant effect on the performance of ejector system. The internal heat exchanger allows lower inlet temperature to the motive nozzle which makes the whole system and the ejector more efficient. Yari and Sirousazar [58] reported on average 7.5–28% higher COP and second law efficiency than those of the conventional ejector-expansion CO_2 refrigeration cycle.

4. Multi-stage compression cycle with ejector

4.1. Subcritical two-stage compression cycle

Works on ejector expansion multi-stage compression cycles are limited. The ejector expansion two-stage compression cycle with intercooling, analyzed by Yari and Sirousazar [59], is shown in Fig. 4. As shown, an internal heat exchanger is also used to get superheated vapor at the compressor entry (state 6) and subcooled liquid as motive stream to ejector. An intercooler is used between the first stage to second stage compressors (process 6a–6b). The flow at state 3 enters the ejector motive nozzle and saturated secondary vapor stream (state 8) enters the ejector suction nozzle. The two streams mix at constant pressure in the ejector and then flows through the ejector diffuser to recover the pressure (state 5). The constant pressure mixing model, discussed above, was used for the theoretical analysis with some selected refrigerants, such as ammonia, isopropanol, propylene, R11, R125, R134a, R141b, R143a, R152a, R22, R290, R32, R500, R502, and R507a. Results [59] showed that the entrainment ratio, COP and second law efficiency of the ejector-expansion two-stage refrigeration cycle are on average 27%, 8.6% and 8.15% higher than that of the conventional ejector expansion refrigeration cycle. Also, the COP and second law efficiency of the vapor compression cycle with intercooler and internal heat exchanger are 21% and 8.1%, respectively, higher than that of the conventional vapor compression cycle. Among the fluids considered, the best performance improvement in the COP was obtained with isopropanol.

4.2. Transcritical two-stage compression cycle

The transcritical CO_2 ejector expansion two-stage compression cycle with intercooler and internal heat exchanger, analyzed by Yari and Sirousazar [60] and Yari [61], is shown in Fig. 5. As shown, the cycle is similar to the subcritical cycle [59] analysed by same authors. Their results based on the first and second laws of thermodynamics showed that use of ejector is more profitable for transcritical cycle: COPs of the new ejector-expansion cycle are 55.5% and 26% higher than that of the conventional CO_2 cycle and conventional ejector-expansion CO_2 cycle, respectively, and the second law efficiencies of the new ejector expansion cycle are 55.5% and 26% higher than that of the conventional CO_2 cycle and conventional ejector-expansion CO_2 cycle, respectively, at evaporator temperature of 10 °C, gas cooler outlet temperature of 40 °C and optimum gas cooler pressure. Second law optimization [61] showed that the COP and second law efficiency of the ejector-expansion two-stage CO_2 cycle are on average 16.5%, 18.4% and 28.4% higher than that of the two-stage CO_2 cycle with internal heat exchanger and intercooler, the two-stage CO_2 cycle with flash gas bypass, and the two-stage CO_2 cycle with flash gas intercooling cycles, respectively, while the optimum inter-stage pressure is approximately equal to geometric mean pressures of evaporator and gas cooler. The optimum gas cooler pressure decreases by the use of ejector in two-stage compression with intercooling [62]. It may be noted that the performance may be further improved by using multi-intercooling [63]. Cogswell et al. [34] recently patented ejector expansion two-stage compression cycle with intercooling, and with and without internal heat exchanger.

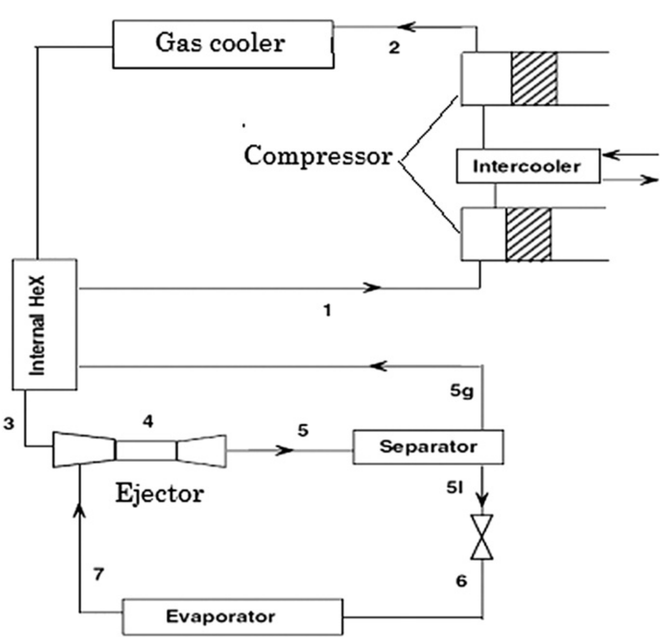

Fig. 5. Ejector expansion multistage compression transcritical cycle.

5. Multi-evaporator cycle with ejector

5.1. Multi-evaporator subcritical cycle

Tomasak and Radermacher [64] and Elakdhar et al. [65] proposed a compression–ejection hybrid cycle for domestic refrigeration to reduce the loss of available energy due to the large temperature difference between the fresh food section and the freezer section. This system employs eight components: a compressor, an ejector, a condenser, a separator, two evaporators, and two capillary tubes as shown in Fig. 6(a). The refrigerant vapor at low pressure is compressed to the condenser pressure (1–2) and then enters the condenser where it condenses to state 3 by rejecting heat to the surroundings. The condensate enters the evaporator 1 via the capillary tube 1 (3–4). The two phase refrigerant from evaporator 1 (state 5) is separated into two tributaries: the vapor refrigerant (state 6) flows to the ejector as motive stream, and the liquid refrigerant (state 7) flows to the capillary tube 2. Thus, the two evaporators have different matching evaporation temperatures with the fresh food section and freezer section. The liquid fluid entering the evaporator 2 is vaporized from state 8 to state 9 and produces a cooling effect by absorbing heat from the refrigerated space. The higher pressure saturated vapor refrigerant (state 6) is used to compress the refrigerant vapor leaving the evaporator 2 (state 9), and it increases the suction pressure of the compressor and decreases the compressor pressure ratio. On-dimensional constant pressure mixing ejector model was used for performance evaluation using several refrigerants (R123, R124, R141b, R290, R152a, R717, R600a, and R134a). Theoretical study [65] showed that entrainment ratio and the coefficient of performance depend mainly on the fluid nature and the operating conditions. It was found that the hybrid cycle has a significant gain in COP and needs less mechanical work as compared with the simple vapor compression cycle. For the same operating temperatures of the ejector refrigeration systems, R141b gives the most advantageous relative COP. It was determined that performances increased with increasing the fresh food evaporating temperature and decreased with increasing the condenser and the freezer evaporating temperatures. The system performance can be improved by 32% relative to the conventional cycle. It may be noted that if the high temperature evaporator load is zero, then the cycle becomes a booster vapor-compression refrigerating system proposed by Buyadgie et al. [66]. Their calculations have shown that COP of proposed system is 10–15% higher comparing to conventional ejector expansion vapor compression system.

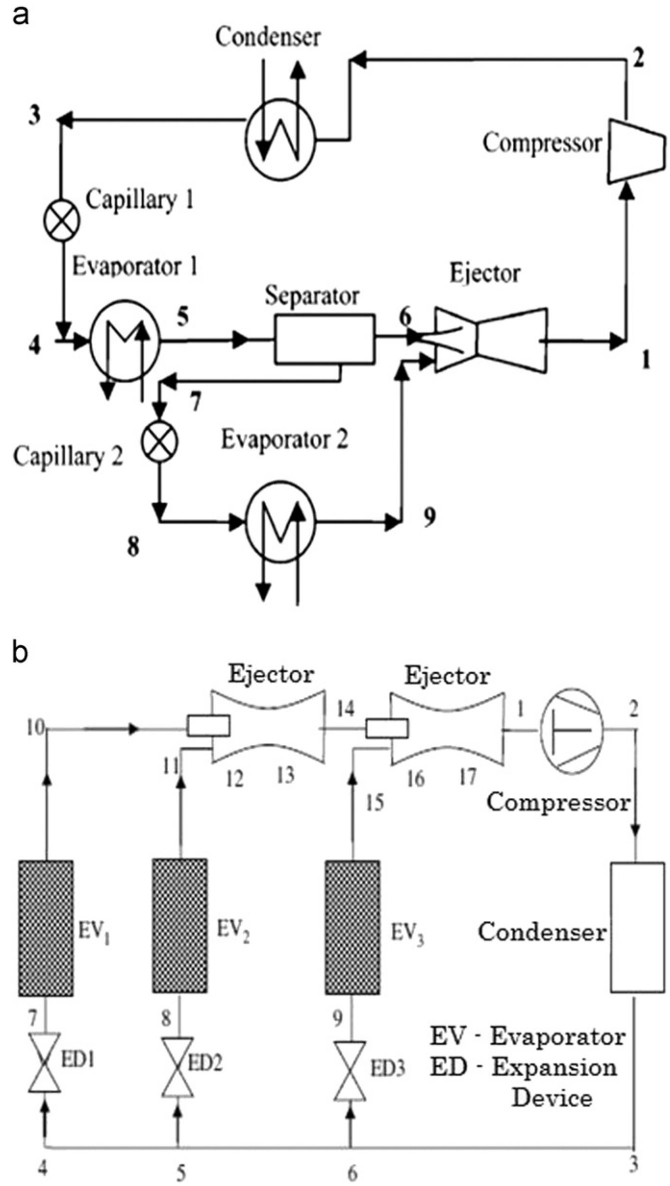

Fig. 6. Ejector expansion multi-evaporator subcritical cycle.

Kairouani et al. [67] extended their analysis for three evaporator system using two ejectors (Fig. 6(b)). After compression (1–2) and heat rejection (2–3), the condensate is divided into three flows, states 4, 5 and 6. One enters the evaporator 1 after a pressure reduction in the expansion device 1, state (7), the other enters the evaporator 2 after a pressure reduction in the expansion device 2, state (8) and the last enters the evaporator 3 after a pressure reduction in the expansion device 3, state (9). After the evaporator 1 (7 −1 0), the superheated vapor enters through motive nozzle and after the evaporator 2 (8 −1 1), the superheated vapor enters through secondary nozzle of ejector 1. The mixed fluid at the exit of ejector 1 (state 14) again enters the ejector 2 as motive stream and compress the superheated refrigerant vapor leaving the evaporator 3 (state 15), and it increases the suction pressure of the compressor (state 1) and decreases the compressor pressure ratio. 1-D constant-area ejector flow model of the ejector including effects of friction at the constant area mixing chamber was used for performance analysis with environment friendly refrigerants (R290, R600a, R717, R134a, R152a, and R141b). Their theoretical study [67] showed that the COP of the novel refrigeration cycle with three evaporators and two ejectors is better than the conventional system. For the same operating temperatures, within the refrigerants considered [67], R141b gives the most advantageous relative coefficient of performance. Lin et al. [68] investigated the effects of varying cooling loads on performances of ejector expansion three-evaporator refrigeration system (Fig. 6(b)) by CFD technique using R134a as refrigerant. Their results indicated that pressure recovery ratio is very sensitive to the varying primary and secondary flow cooling loads. The maximum pressure recovery ratio can reach 60% as the cooling loads vary. It was found that in order to keep the system stable, the primary and secondary cooling loads should be maintained within $\pm 5\%$ and $\pm 10\%$, respectively, in which case the pressure recovery ratio will have a maximum ratio of 32.8%.

5.2. Multi-evaporator transcritical cycle

Sarkar [69] proposed two novel layouts of multi-evaporator transcritical CO_2 refrigeration system using constant pressure mixing ejector as shown in Fig. 7(a) and (b). Both cycles consist of single compressor and single gas cooler. However, in the first cycle, the fluid at state 3 enters evaporator 1 via expansion device 1 and then separated into two tributaries: the vapor refrigerant at state 6 flows to the ejector, and the liquid refrigerant at state 14 flows to expansion device 2 (14–15) and mixes with liquid from separator 2 via expansion device 3 and enters the evaporator 2. The primary flow (state 6) and the secondary flow (state 7) are going through primary and secondary nozzles, respectively, to the

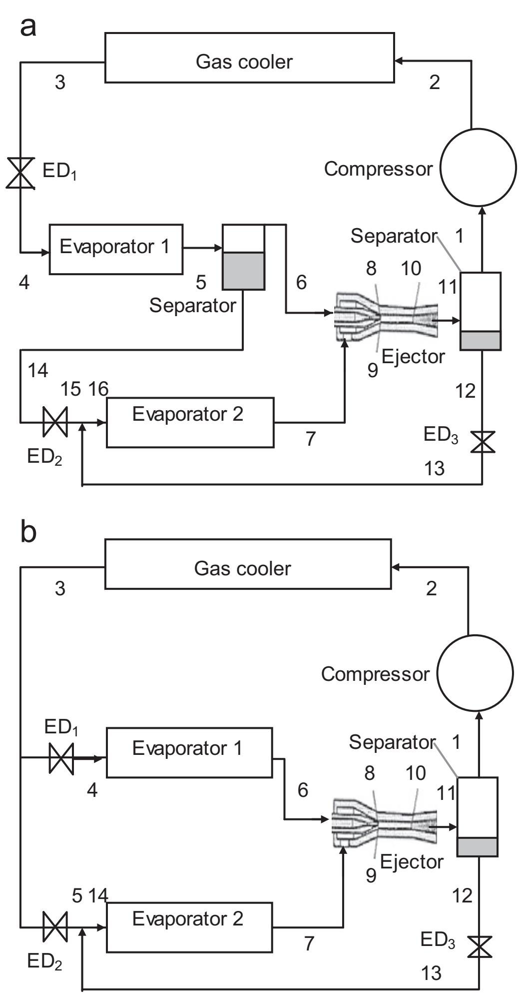

Fig. 7. Ejector expansion multi-evaporator transcritical cycle.

constant pressure mixing and diffuser (states 10 and 11) sections of the ejector and then separated in forms of vapor (state 1) and liquid (state 12). Whereas, in second cycle, fluid after gas cooler flow through evaporator 1 and evaporator 2 separately via individual expansion devices and the saturated vapor leaving evaporator 1 is used as the ejector motive fluid (state 6) to compress the refrigerant vapor leaving evaporator 2 (state 7), and it increases the suction pressure of the compressor. The refrigerant exiting from ejector (state 11) separates in forms of vapor and liquid: the vapor flows through compressor and liquid enters the evaporator 2 after expanding through expansion valve.

Study [69] showed that the performance for both cycles increases with the increase in cooling capacity ratio of evaporator 1 to evaporator 2. The optimum discharge pressure for both cycle layouts compared to corresponding basic cycle reduces by the use of ejection–compression. For both cycle layouts, the effect of gas cooler exit temperature on COP improvement is negligible, whereas the COP improvement at optimum condition increases with the increase in temperature difference between evaporators. Results revealed that the first one (Fig. 7(a)) is superior due to lower discharge pressure and significantly higher COP improvement (maximum of 38.1% whereas 21.4% for second one (Fig. 7(b)) at optimum condition for the studied ranges). However, the cost may be slightly higher due to the use of an extra separator.

6. Miscellaneous cycle configurations with ejector

Yu et al. [70] proposed a novel autocascade refrigeration cycle with an ejector as shown in Fig. 8. The ejector is set between the evaporative condenser and the evaporator. After compression (1–2), the refrigerant mixture is partially condensed and two phase stream flows into the phase separator, from which the liquid flows through the throttling device 1 and enters the evaporative condenser at a lower temperature (at point 7) to cool the vapor phase stream from the phase separator and then enters the ejector as primary flow after vaporizing. The vapor from the phase separator condenses and after expanding this subcooled liquid enters the evaporator to give refrigerating effect. Then, the vapor is sucked by the ejector and then the mixed vapor is discharged at high suction pressure. By this way, the irreversible loss of throttling process in this throttling device will be reduced and the partial usable work can be recovered to lift the evaporator outlet pressure. Theoretical computation model based on the constant pressure mixing model for the ejector was used to perform a thermodynamic cycle analysis with the refrigerant mixture of R23/R134a. The results show the ejector cycle has an outstanding merit in decreasing the pressure ratio of compressor as well as increasing the COP. At the condenser outlet temperature of 40 °C, the evaporator inlet temperature of −40.3 °C, and the mass fraction of R23 is 0.15, the pressure ratio of compressor is reduced by 25.8% and the COP is improved by 19.1% over the conventional cycle.

Oshitani et al. [71] developed an ejector type cool box system to achieve continuous operation with two evaporator temperatures in the same system for vehicle air-conditioning systems. As shown in Fig. 9(a), an air-conditioning evaporator is used between ejector and separator; otherwise the system is similar to conventional ejector expansion refrigeration cycle. They successfully commercialized the ejector system cool box, which provides substantial performance improvements at a level unachievable using a conventional system (Cabin air-conditioning: +15% in performance; Refrigeration: 20% reduction in cooling time). Gocho et al. [72] proposed a new automotive ejector type cool box system for cooling beverages especially when driving in hot

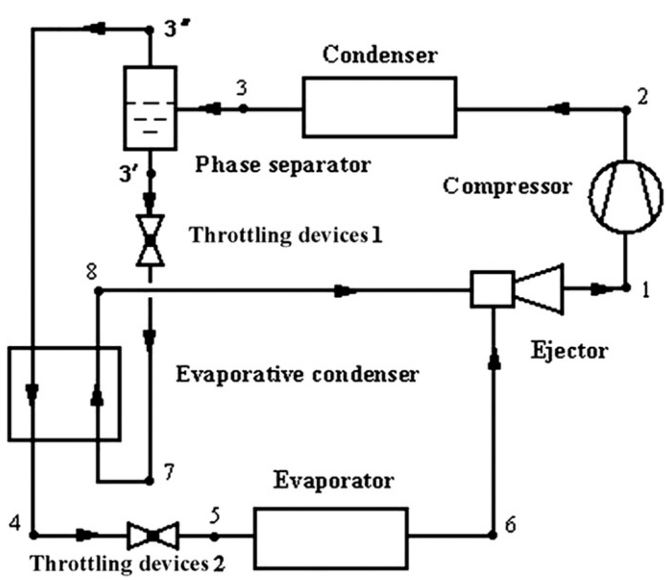

Fig. 8. Autocascade refrigeration cycle with an ejector.

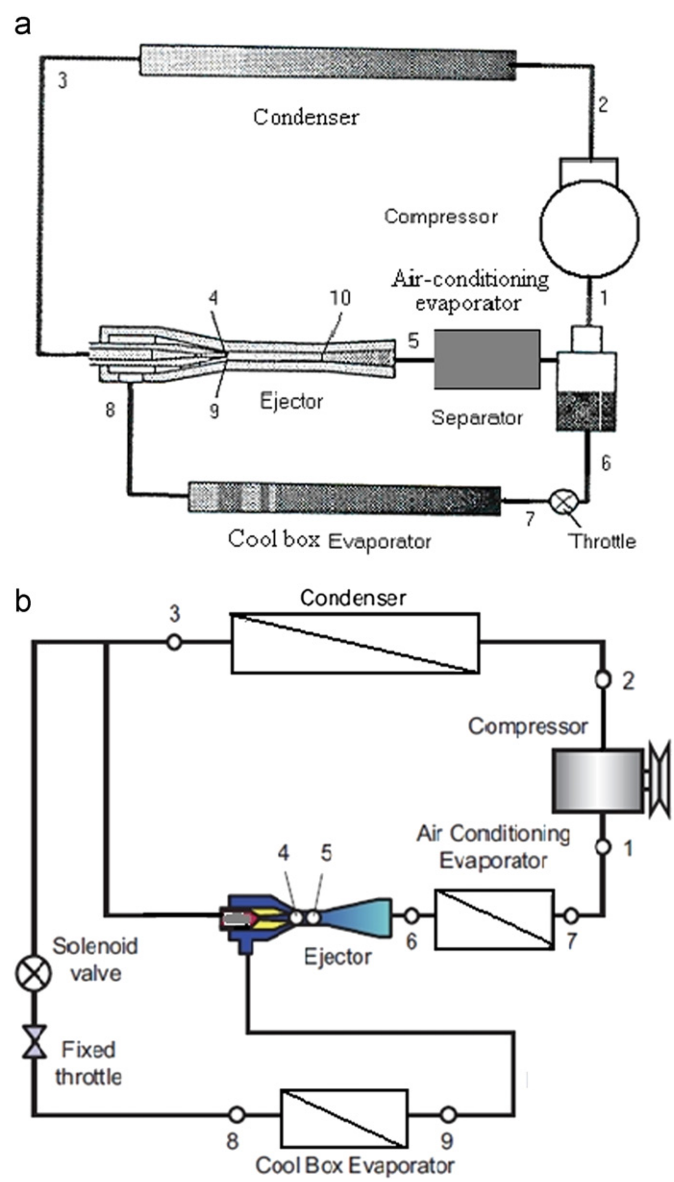

Fig. 9. (a) Ejector type cool box system and (b) Modified ejector-type cool box system.

regions. As shown in Fig. 9(b), the flow after condenser is divided into to parts: one part, after expanding by expansion device flows through cool box evaporator and is used as suction flow whereas another part is used as motive flow in ejector. After ejector, fluid is flows air-condioning evaporator and then compressor and condenser. By this way, cool box is maintained at lower temperature than the vehicle cabin. Consiquently, significant performance improvements have been achieved for the whole vehicle refrigeration system.

Xu and Ma [73–75] used ejector in quasi two-stage compression heat pump system coupled with vapor injection compressor (economizer) and ejector, as shown in Fig. 10. Compared with the conventional system, the throttling valve in the supplementary circuit is replaced by an ejector in the improved heat pump system. As shown, the saturated vapor separated from flash-reservoir, a type of economizer, is used as motive flow and some amount of refrigerant vapor after evaporator is taken as secondary flow in ejector. Both streams mix and then the pressure of the mixing vapor rises to the intermediate pressure, which is sucked into the compressor through supplementary inlets and the flow path is called the supplementary circuit. By this way, instead of compressing from evaporator pressure, some part of refrigerant compresses from intermediate pressure (higher than evaporator pressure) and hence the compressor work reduces leading to COP improvement. The ejector can reduce the available energy loss, and the temperature of the refrigerant vapor entering into the scroll compressor through supplementary inlets, so it can improve the compression process of the heat pump system further. They have done the experimental analysis with refrigerant R22. They demonstrated that the heating energy efficiency ratio could reach about 4% higher than that of the system without the ejector when the heating capacity remained nearly constant [73,74]. Xu and Ma [75] showed that the ejector can effectively improve the performance of the heat pump, and the exergy efficiency could be increased 3–5% by replacing the throttle valve in the supplementary circuit of the quasi two-stage heat pump with an ejector, while the rate of exergy output would not decrease.

Dopazo and Seara [76] experimentally evaluated the performance of an ejector working as a liquid re-circulator component in an overfeed plate evaporator with NH_3. The experimental facility consists of a cascade refrigeration system, where NH_3 system is connected with the CO_2 based low temperature single stage system (Fig. 11). If the evaporator (cascade heat exchanger) exit condition is wet (two-phase), the liquid is separated and re-

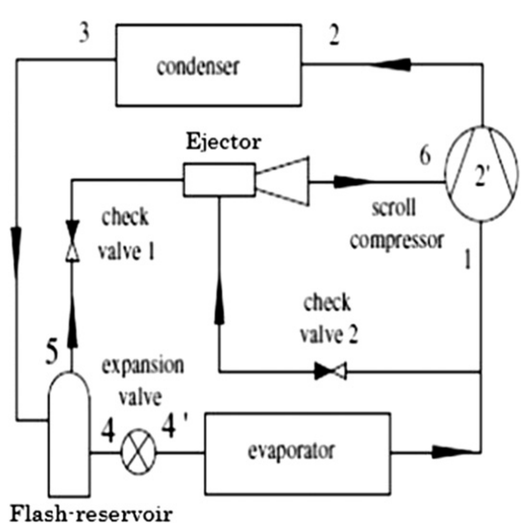

Fig. 10. Heat pump with economized compressor and ejector.

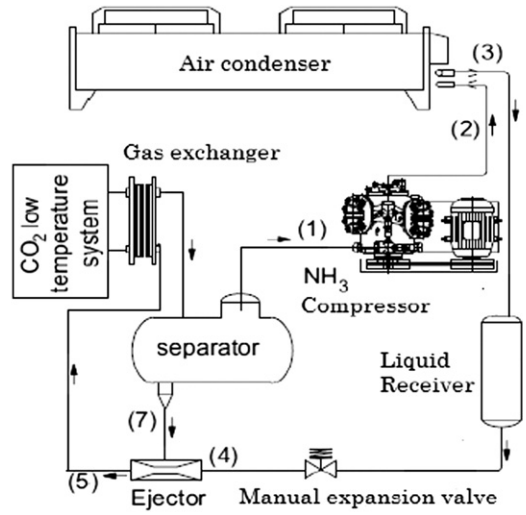

Fig. 11. Use of ejector as liquid re-circulator in an overfeed NH_3 system.

circulated to evaporator again through the ejector. The experimental results showed that, as the main liquid mass flow rate increases, the re-circulated liquid volumetric flow rate tends to remain constant.

Chen et al. [77] studied a novel vapor compression heat pump cycle in which an ejector associated with a subcooler is applied to enhance the heating performance for air-source heat pump water heater application (Fig. 12). In this cycle, the high-pressure vapor from the compressor is as the motive stream to entrain the vapor (secondary fluid) with the subcooler pressure. The refrigerant vapor leaving the ejector enters the condenser where it condenses by rejecting heat to get desired heating and separates into two streams: one stream flows through the expansion valve I and enters the subcooler, where it is evaporated and entrained into the ejector again, and the other stream is further cooled in subcooler and enters the evaporator through expansion valve II. The results showed that the COP and volumetric heating capacity of the novel cycle are better than that of the conventional heat pump cycle (maximum improvements of 6.92% and 37.32%, respectively, for the operating conditions considered).

Wang et al. [78] proposed a new cooling, heating and power (CCHP) system to produce cooling output, heating output and power output simultaneously. This proposed combined system combines a Brayton cycle and a transcritical CO_2 refrigeration cycle with ejector as expansion device, which uses solar energy as the heat source. As shown in Fig. 13, the sub-critical CO_2 is compressed into the compressor to a supercritical state. The supercritical CO_2 is heated in gas heater using solar heating system and expanded to a low supercritical pressure in the turbine to produce power. Then it enters the heater to supply the heat to the heating user, and then rejects heat in the gas cooler. The flow coming from the gas cooler is used as the primary flow to drive an ejector to entrain secondary vapor from the evaporator. The CO_2 stream leaving the ejector flows into the separator where it is divided into saturated liquid and saturated vapor streams. The saturated liquid enters the evaporator through a throttle valve, and a part of the saturated vapor is also fed back to the evaporator through a throttle valve. The CO_2 in the evaporator is vaporized by absorbing heat to produce the cooling effect. The results indicated that increasing turbine inlet pressure and ejector inlet temperature could lower the efficiency of the system, and increasing turbine back pressure and turbine inlet temperature could elevate the efficiency of system. In addition, as ejector back pressure increases, the thermal efficiency of system decreases, but the exergy efficiency increases.

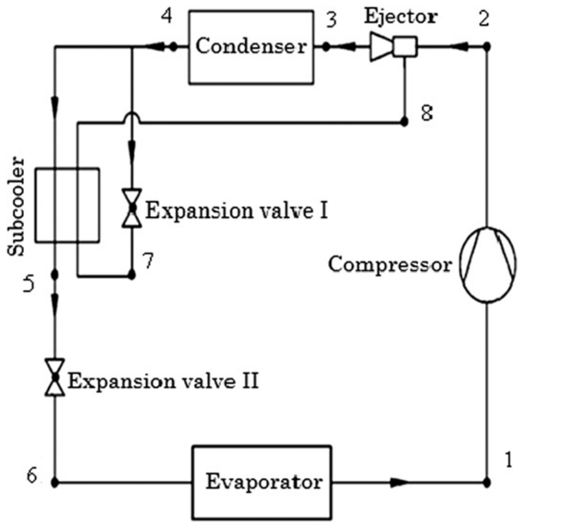

Fig. 12. Ejector enhanced heat pump cycle with subcooler.

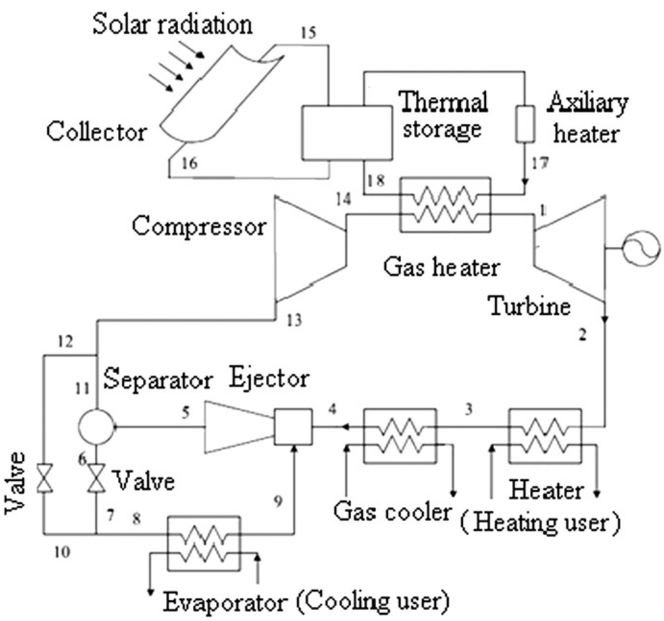

Fig. 13. Ejector enhanced cooling, heating and power system.

7. Conclusions

The two-phase ejectors and their applications in vapor compression refrigeration/heat pump system have been reviewed and the various ejector enhanced system configurations and their performance characteristics are well-documented here. The effects of various operating parameters, geometric parameters and refrigerant varieties on the ejector performances (entrainment ratio, pressure lift ratio and ejector efficiency) as well as system performances (system capacity, COP, exergy loss or irreversibility and second law efficiency) have been well-summarized for various subcritical and transcritical cycle configurations. Ejector geometry, operation, modeling and performance are discussed in details. Almost all theoretical studies on ejector are based on one-dimensional homogeneous equilibrium model with fixed isentropic efficiencies of nozzle and diffuser. However, more realistic analysis of vapor compression cycle with the two-phase ejector can be done by using multi-dimensional non-homogeneous flow including the friction factor and shock calculations.

Both theoretical and experimental studies showed that the performance of subcritical as well as transcritical refrigeration and heat pump cycles can be improved significantly by using ejector as an expansion device. The energetic or exergetic performance improvement by using ejector is strongly dependent on cycle operating conditions, working fluids used and ejector geometries. Various ejector geometric parameters such as dimensions of motive nozzle, suction nozzles, mixing sections and diffuser significantly affect the ejector as well as cycle performances. Furthermore the geometric parameters can be adjusted at the optimum level during the cycle operation to get the maximum ejector as well as system performances. Optimum gas cooler pressure for transcritical cycle can be also maintained by adjusting the ejector geometry.

Use of ejector in vapor compression system not only improves the performance also simplify operation and control at optimum level. It is hoped that this contribution will stimulate wider interest in the technology of two-phase ejectors and their applications in vapor-compression system. It should be useful for commercial prototype development and any newcomer in this field of technology.

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